IS 11639-1 (1986): Criteria for structural design of penstocks, Part 1: Surface penstocks [WRD 14: Water Conductor Systems]

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(Reaffirmed 2001)

Indian Standard

CRITERIA FOR STRUCTURAL DESIGN OF PENSTOCKS

PART 1 SURFACE PENSTOCKS

UDC 627·844 : 624·04

INDIAN STANDARDS INSTITUTION
MANAK BHAVAN, 9 BAHADUR SHAH ZAFAR MARG
NEW DELHI 110002

Gr 4 *November* 1986

AMENDMENT NO. 1 DECEMBER 2008
TO
IS 11639 (PART 1) : 1986 CRITERIA FOR STRUCTURAL DESIGN OF PENSTOCKS

[*Page* 9, *clause* **6.1.5.1**(b)] — Substitute '$f_3 = \frac{\mu_1 P A_c}{A}$' *for* '$f_3 = \mu_1 P A_c$'.

(*Page* 11, *clause* **7.1**) — Substitute '$S_x, S_y = \frac{f_x + f_y}{2} \pm \sqrt{\left(\frac{f_x - f_y}{2}\right)^2 + q^2}$' *for* '$S_x, S_y = \frac{f_x + f_y}{2} \pm \sqrt{\frac{(f_x - f_y)^2}{2} + q^2}$'.

(WRD 14)

Reprography Unit, BIS, New Delhi, India

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CRITERIA FOR STRUCTURAL DESIGN OF PENSTOCKS
PART 1 SURFACE PENSTOCKS

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Indian Standard

CRITERIA FOR STRUCTURAL DESIGN OF PENSTOCKS

PART 1 SURFACE PENSTOCKS

0. FOREWORD

0.1 This Indian Standard (Part 1) was adopted by the Indian Standards Institution on 31 January 1986, after the draft finalized by the Water Conductor Systems Sectional Committee had been approved by the Civil Engineering Division Council.

0.2 Conduits carrying water from surge tanks or directly from a reservoir, forebay, to the power house are known as penstocks. Penstocks are generally of concrete or steel or a combination of both. The pressure varies from minimum at the upstream end to the maximum at the junction with the scroll case.

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1. SCOPE

1.1 This standard (Part 1) lays down the various forces acting on surface penstocks and structural design of penstocks conveying water under pressure flow conditions. However, this does not cover specials of penstocks like penstock supports, manifolds, bends, expansion joints, manholes, branch outlets, etc.

2. NOTATIONS

2.1 For the purpose of this standard, the following notations shall have the meaning indicated against each:

A = cross-sectional area of pipe shell material, mm^2

A_r = cross sectional area of stiffener ring, mm^2

b = width of ring girder or stiffener ring, mm

C = moment coefficient

E = modulus of elasticity

f_1, f_2, f_3 = longitudinal stresses, N/mm^2

f_4 = secondary bending stress, N/mm^2

f_x = total circumferential stress, N/mm^2
f_y = total longitudinal stress, N/mm^2
l = *height of stiffener ring or ring girder, mm*
L = span length of pipe, mm
M, M_1 = moments
P = internal pressure including water hammer, N/mm^2
P_1 = total reaction at support, N
q_1 = shear stress, N/mm^2
r = radius of pipe shell, mm
r_1 = mean radius of shell, mm
S = hoop stress in pipe, N/mm^2
S_e = equivalent stress, N/mm^2
S_x, S_y = principal stresses, N/mm^2
T = temperature rise or drop, °C
t = thickness of pipe shell, mm
t_1 = thickness of stiffener ring or ring girder, mm
W = total distributed weight, that is, self-weight of shell + weight of water, N/m^2
W_1 = total weight, that is, weight of shell + weight of water, N
Z = section modulus of pipe shell, m^3
α = coefficient of linear expansion or contraction of pipe shell material, per °C
μ = coefficient of friction.

3. DATA REQUIRED

3.1 The following data is required for the structural design of penstocks:

a) General drawing of installation;
b) Complete longitudinal profile;
c) Type of penstock;
   i) Free penstock pipe laid in a tunnel, and
   ii) Steel lined pressure shaft.
d) Geological data;
   i) Geology of area,
   ii) Type of ground/rock,
   iii) Bearing capacity of ground,

iv) Shear properties of soil,

v) Modulus of deformation, and

vi) Seismic coefficient.

e) Climatic conditions;

i) Temperature (maximum, mean, minimum, by day and night and in summer and winter),

ii) Wind conditions (direction and maximum speed), and

iii) Snow conditions (period and average depth of snow).

f) Hydraulic data;

i) Diameter of penstock, and

ii) Discharge through penstock.

4. LOADS ON PENSTOCK

4.1 The following are the main loads considered for the design of penstocks:

a) Internal water pressure,

b) Weight of penstock and water, and

c) Temperature.

In addition, the other loads considered depending on location are:

a) wind load,

b) snow load, and

c) seismic forces.

4.2 The loads given in **4.1** are of the following nature:

a) Permanent loads,

b) Intermittent loads, and

c) Exceptional loads.

4.2.1 The loads of permanent nature are the forces which act upon the penstock in normal operation. They correspond to:

a) The maximum operating pressure which is the sum of maximum static pressure and over pressure due to water hammer under normal operating conditions, taking into account the oscillations in surge tank;

b) The weight of penstock and water between the supports;

c) Spacing and type of supports;

d) The difference between the temperature which may exist in the penstock in normal operation and the temperature existing when coupling up the sections during erection;

e) Friction at supports; and

f) Temperature variation.

4.2.2 The loads of intermittent nature are the forces which though not exceptional, do not arise often.

The two main cases are:

a) Penstock during filling, and

b) Empty penstock under partial vacuum.

The forces to be taken into account in each case correspond to:

a) The weight of penstock and water between the supports,

b) The type and spacing between supports,

c) The temperature effect, and

d) Friction at supports.

In addition to the forces enumerated in (a) to (d), intermittent loads may also occur due to wind or snow load and earthquakes.

4.2.3 The loads of exceptional nature that may act upon the penstocks are:

a) Shop or site test;

b) Erection stresses;

c) Bad operation of safety devices during filling such as non-operation of air valves which would create vacuum inside during empting operation of penstocks;

d) Pressure rise caused due to unforeseen operation of regulating equipment of turbine/pump distribution:

   i) In case of impulse turbine, pressure rise due to needle slam on loss of oil pressure or mechanical failure; and

   ii) In case of reaction turbine, turbine gates may be closed instantaneously at any time by action of governor manual control of main relay valve or by the emergency solenoid device.

e) Stresses developed due to resonance in penstock (As far as possible, the frequency of penstock pipe in any reach shall not match with the frequency of machine, frequency of vortex shedding in the draft tube, frequency of the system, etc, and resonance shall be avoided); and

f) Seismic forces.

5. STRESSES IN PENSTOCK SHELL

5.1 The stresses in the penstock shell of surface penstocks are subjected to circumferential and longitudinal stresses. The stresses in pipe at the mid-span and at the supports are as below:

a) *At mid-span*:

   i) Hoop stresses developed due to internal pressure equal to sum of static pressure due to maximum water level in reservoir or surge tank plus the dynamic pressure due to water hammer as calculated for operating conditions,

   ii) Longitudinal stresses developed due to its own weight and weight of water by beam action,

   iii) Longitudinal stresses developed due to sliding friction over the supports, and

   iv) Longitudinal stresses developed due to expansion or contraction of penstock shell due to variation of temperature.

b) *At supports*:

   i) Circumferential stresses developed at the supports due to bending caused by internal pressure,

   ii) Longitudinal stresses due to secondary bending moments caused by the restraints imposed by ring girder or stiffener rings,

   iii) Longitudinal stresses developed at the supports due to beam action, and

   iv) Longitudinal stresses developed by forces enumerated in **4.1**.

5.2 The stresses in penstock shell shall also be checked to withstand the stresses developed due to intermittent and exceptional loading and for the following forces:

a) Longitudinal stresses developed due to earthquake and wind,

b) Circumferential stresses developed due to pressure rise called by non-operation as specified in **4.2.3** (d),

c) Longitudinal stresses developed due to wind, and

d) Stresses developed due to filling and draining of penstocks.

6. METHOD OF CALCULATION OF STRESSES IN PIPE SHELL

6.1 The stresses for different forces shall be calculated as given in **6.1.1** to **6.1.7**.

6.1.1 *Hoop Stress Due to Internal Pressure*

6.1.1.1 The hoop stress developed due to internal pressure is given by:

$$S = \frac{Pr}{t}$$

NOTE — The internal pressure P is due to static head + dynamic head.

6.1.2 *Longitudinal Joint Efficiency (e)*

6.1.2.1 It Shall be taken as 1·0 for fully radiographed joint.

6.1.3 *Longitudinal Stresses Due to Beam Action*

6.1.3.1 The stress developed due to self-weight and weight of water pipe spanning over supports due to beam action shall be calculated by the following formula:

$$f = \frac{M}{Z} \text{ N/m}^2$$

where

M = bending moment caused due to self-weight and weight of water.

NOTE — Bending moment, M, at mid-span and at support shall be calculated for each case considering the pipe as a beam spanning continuously over intermediate supports

6.1.3.2 The longitudinal stresses developed due to radial strain caused by internal pressure may be taken equal to 0·303 times the hoop tension.

6.1.3.3 The longitudinal stresses developed due to seismic forces in the pipe spanning over the support due to beam action is equivalent to seismic coefficient times stresses calculated in **6.1.3.1**. These stresses should be added to stresses calculated in **6.1.3.1**.

6.1.3.4 The longitudinal stresses developed due to wind or snow load acting on the pipe spanning over the support are calculated as given in **6.1.3.1**, where bending moment, M, is caused due to wind load or snow load.

6.1.4 *Longitudinal Stress Due to Sliding Friction*

6.1.4.1 The maximum longitudinal stress developing over all supports within a section between an expansion joint and the subsequent support shall be calculated by the formula:

$$f_2 = \frac{\Sigma Pf}{A} + \frac{a\,\Sigma Pf}{Z},$$

where

Pf = total sliding friction in section between an expansion joint and subsequent support = μW, Cos β,

a = eccentricity of frictional force relative to centre line of penstock,

Z = section modulus of pipe shell, and

β = angle of section under consideration with the horizontal.

The coefficient of friction, μ, between shell and support shall be taken from Table 1.

TABE 1 FRICTION COEFFICIENTS FOR DIFFERENT MATERIALS

(*Clause* 6.1.4.1)

Sl No.	Type of Support	Friction Coefficient
i)	Steel on concrete	0·60
ii)	Steel on concrete with asphalt roofing paper in between	0·50
iii)	Steel on steel (rusty)	0·50
iv)	Steel on steel (greased)	0·25
v)	Steel on steel with two layers of graphite service sheets in between	0·25
vi)	Rocker support	0·15
vii)	Roller support	0·10
viii)	Concrete on concrete	0·75

6.1.5 *Temperature Stress*

6.1.5.1 Longitudinal stresses caused due to expansion or contraction of pipe shell shall be calculated by the following formulae:

a) Pipe shell without expansion joint

$$f_3 = E \alpha T$$

b) Pipe shell with expansion joint

$$f_3 = \mu_1, PAc,$$

where

μ_1 = coefficient of friction between packing material and pipe shell, and

Ac = contact area between penstock shell and packing material in mm^2.

6.1.6 *Circumferential Bending Stresses at Supports*

6.1.6.1 Circumferential stresses at supports due to bending caused by internal pressure shall be calculated by the following formula:

$$M = C P_1 r$$

The value of C for different angles of support is given in Fig. 1. The effective length of shell resisting the bending moment is equal to 4 times the radius of the shell.

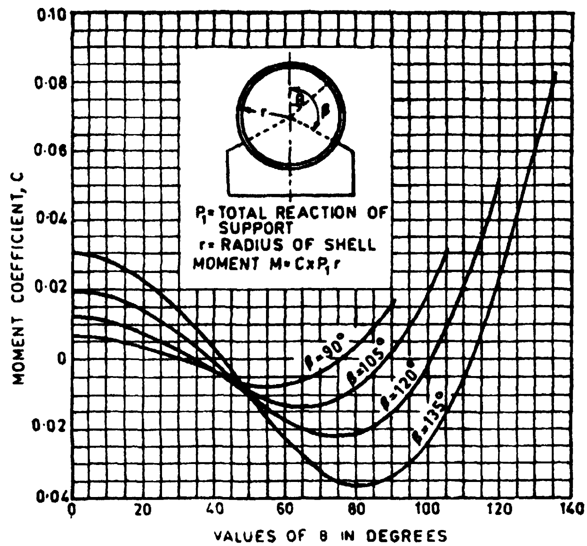

FIG. 1 VARIATION OF CIRCUMFERENTIAL MOMENT AROUND SHELL RESTING ON SADDLE SUPPORT

6.1.7 *Longitudinal Bending Stress Due to Restrain by Ring Girders or Stiffener Rings at the Supports*

6.1.7.1 The secondary bending stress due to restrain shall be calculated by the formula:

$$f_4 = \frac{1{\cdot}82\,(\,Ar - bt\,)}{Ar + 1{\cdot}56\,t\sqrt{rt}} \times \frac{Pr}{t}$$

NOTE — This stress is local and the effect shall be taken for a distance of $3/q$ where $q = \frac{1{\cdot}285}{\sqrt{rt}}$ on either side of the supporting ring and pipe thickness shall be increased, if required, in this zone.

7. EQUIVALENT STRESSES

7.1 The circumferential and longitudinal stresses obtained as specified in **6.1.1** to **6.1.7**, shall be combined to obtain equivalent stresses in accordance with Hencky Mises Theory, given by the formula:

$$S_e = \sqrt{S_x^2 + S_y^2 \pm S_x S_y}$$

$$S_x, S_y = \frac{f_x + f_y}{2} \pm \sqrt{\frac{(f_x - f_y)^2}{2} + q^2}$$

where

f_x = longitudinal stress,

f_y = hoop stress, and

q = shear stress

The equivalent stresses may be readily obtained from Fig. 2.

8. LINER THICKNESS

8.1 The liner thickness of an exposed penstock shall withstand the equivalent stress as specified in **7**.

8.2 Notwithstanding the thickness obtained as specified in **8.1** and regardless of pressure, a minimum thickness of liner shall be provided to resist the distortion during fabrication and erection. A minimum thickness of $\frac{D + 50}{400}$ cm is recommended where D is the diameter of shell in cm.

8.3 No corrosion allowance is recommended. Instead, it is suggested to paint the inside and the outside surface of pipe with a paint conforming to the relevant Indian Standard.

9. WORKING STRESSES AND FACTOR OF SAFETY

9.1 Normal Operating Condition

9.1.1 It is recommended that under normal operating condition as specified in **4.2.1**, the working stresses with a factor of safety of 3 based on the minimum ultimate tensile strength shall be adopted for design but in no case the maximum stresses obtained in **7.1** shall exceed 0·5 times the specified minimum yield strength of material.

9.2 Intermittent Loading Condition

9.2.1 It is recommended that under intermittent loading condition as specified in **4.2.2**, the working stresses with a factor of safety of 2·5 based on minimum ultimate tensile strength shall be adopted for designs but in no case the maximum stresses obtained in **7.1** shall exceed 2/3 the specified minimum yield strength of material.

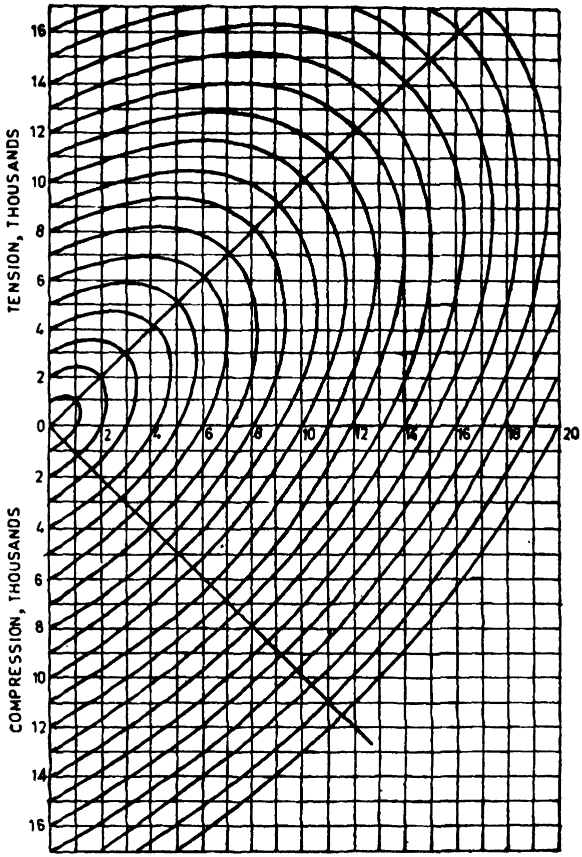

FIG. 2 EQUIVALENT STRESS DIAGRAM

9.3 Exception Condition

9.3.1 Under exceptional loading condition as specified in **4.2.3**, it is recommended that the working stresses with a factor of safety 2·0 based on minimum ultimate strength shall be adopted for design but in no case the maximum stresses obtained in **7.1** shall exceed 0·8 times the specified minimum yield strength of material.

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INTERNATIONAL SYSTEM OF UNITS (SI UNITS)

Base Units

QUANTITY	UNIT	SYMBOL
Length	metre	m
Mass	kilogram	kg
Time	second	s
Electric current	ampere	A
Thermodynamic temperature	kelvin	K
Luminous intensity	candela	cd
Amount of substance	mole	mol

Supplementary Units

QUANTITY	UNIT	SYMBOL
Plane angle	radian	rad
Solid angle	steradian	sr

Derived Units

QUANTITY	UNIT	SYMBOL	DEFINITION
Force	newton	N	1 N = 1 kg.m/s^2
Energy	joule	J	1 J = 1 N.m
Power	watt	W	1 W = 1 J/s
Flux	weber	Wb	1 Wb = 1 V.s
Flux density	tesla	T	1 T = 1 Wb/m^2
Frequency	hertz	Hz	1 Hz = 1 c/s (s^{-1})
Electric conductance	siemens	S	1 S = 1 A/V
Electromotive force	volt	V	1 V = 1 W/A
Pressure, stress	pascal	Pa	1 Pa = 1 N/m^2